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Heat Transfer Coefficient, Two-Phase Flow Boiling of HFC134a

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HEAT TRANSFER COEFFICIENTS AND PRESSURE DROP FOR FORCED CONVECTION BOILING AND CONDENSATION OF HFC 134a

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ABSTRACT

Experimental measurements of local heat transfer coefficients and pressure drop are made for forced convection boiling and condensation for R134a-oil mixtures inside a horizontal tube.

The results are compared with the experimental data of pure R134a. Average convection boiling and condensation heat transfer coefficients decreased about 20% at a 10% oil mass fraction, and boiling and condensation pressure drop were 2.5 and 1.4 times that of pure refrigerant flow respectively.

INTRODUCTION

R134a is most likely to be an alternative refrigerant for R12 which is used as a refrigerant in automotive air conditioners and domestic refrigerators. Therefore, the heat transfer coefficient and in-tube pressure drop for R134a are desired in designing new systems that use R134a as the working fluid. Especially, the influence of oil on the heat transfer coefficient and the pressure drop in heat exchangers have to be estimated, because lubricants always circulate in actual vapor compression refrigeration systems.

Eckels and Pate (1990) estimated the heat transfer coefficients for R134a by using reliable correlations, and compared them to the heat transfer coefficients for R12. However, they did not take into account the oil effect on the heat transfer coefficients.

An experimental study of the heat transfer coefficients and pressure drop for forced convection boiling and condensation of R134a-oil mixtures inside a horizontal tube are described in this paper. The results are compared with the experimental data for a R12-oil mixture.

FORCED CONVECTION BOILING HEAT TRANSFER COEFFICIENTS AND PRESSURE DROP

Experimental Apparatus

A schematic diagram of the apparatus is shown in Figure 1. This apparatus consists of a refrigeration cycle and an oil circulating loop. The main components of the refrigeration cycle are a hermetic compressor, oil separator, condenser, expansion valve, test section and an evaporator. The oil is eliminated from the compressor discharge gas by the oil separator. The refrigerant mass flow rates are controlled by using the expansion valve, and are measured with a turbine-type flow meter.

The oil contained in the compressor shell is taken out from the compressor shell bottom, and transferred to the test section inlet by a gear-type oil pump. The oil mass flow rates are changed by controlling the oil pump rotating speed, and are measured with a oil flow meter.

The refrigerant-oil mixtures flow into the test section where the heat transfer coefficients and pressure drop are measured. The refrigerant contained in the mixture is completely evaporated into the gas in the evaporator, and the refrigerant-oil mixture enters the compressor shell.

Test Section

The test section consists of a main section 840 mm long and two subsections 400 mm long as shown in Figure 2. They are made of copper tubing 5.5 mm in outside diameter, and their inner cross sections are regular squares of 3.5x3.5 mm. Sheathed electrical heaters 1 mm in outside diameters are wound around the tubes in a uniform pitch and AC voltage is supplied from a variable voltage transformer. The T-type (CC) thermocouples for measuring the tube wall temperature are attached at the tube wall at 90 degree distance and at 4 points along the length of the tube. The temperature and pressure of the refrigerant are also measured at the inlet and outlet of the main section.

Experimentation and Results

Testing was conducted by controlling the oil flow rate injected into the test section, while the refrigerant mass velocity and heat flux of the test section were maintained constant. The quality of the main section inlet refrigerant is controlled by using the subsection heater, and the evaporating pressure is controlled by the bypath valve mounted between the compressor outlet and inlet.

A wattmeter is used to obtain the heat flux, and the inside wall temperatures are calculated from the outside temperatures. At the point in the test section where the temperatures of the refrigerant-oil mixtures are not directly measured, they are calculated from the reading of the two temperatures at the inlet and outlet of the test section assuming that the temperature changes linearly between these points.

The oil mass fraction in the liquid refrigerant is measured at the receiver outlet to verify the function of the oil separator. The liquid refrigerant is taken out from the outlet of the receiver and distilled, and the mass fraction of the oil left is less than 0.05%.

The influence of oil on the local heat transfer coefficients is shown in Figures 3 and 4 at a refrigerant mass velocity of 290kg/m^{2}s and heat flux of 10kW/m^2. Figure 3 is the result for the R134a-PAG (polyalkylene glycol) mixture, and Figure 4 is for the R12-mineral oil mixture.

The oil mass fraction is defined as

$$\xi_o = \frac{G_o}{G_r + G_o} \qquad (1)$$

The local heat transfer coefficient is given below:

$$h = \frac{q}{t_w - t_r} \qquad (2)$$

The quality of the refrigerant-oil mixture is defined as the quality of the refrigerant contained in it:

$$x = \frac{i_{rL} - i_{ls}}{h_{fg}} \qquad (3)$$

Here, i_{rL} is the specific enthalpy of the refrigerant at the point of L m from the inlet of the main section, and can be obtained from the energy equation:

$$M_r(i_{rL} - i_{rI}) + M_o c_{po}(t_{oL} - t_{oI}) = qWL + Q_{sub} \qquad (4)$$

where the thermodynamic properties of R134a are calculated by using the data presented by Willson (1988).

As can be seen from Figures 3 and 4, with an increase in quality the heat transfer coefficient of pure refrigerant increases until it nearly reaches x = 0.9 and then sharply falls. However, the heat transfer coefficients of mixtures decreases gradually in the high quality region. In the low quality region, the oil has little effect on the boiling heat transfer.

Figure 5 shows the influence of oil on the average heat transfer coefficient which is found by integrating the local heat transfer coefficients over the whole quality range. With an increase in the oil mass fraction the average heat transfer coefficient of R134a decreases about 20% at an oil mass fraction of 10%. The maximum deviation of the data for R134a from the values for R12 amounts to as much as 10%.

The effect of the oil mass fraction on the average pressure drop of R134a and R12 at a mass velocity of 290kg/m^{2}s is shown in Figure 6. The pressure drop increases linearly with an increase in the oil mass fraction. The specific volume of R134a is larger than that of R12, therefore, the pressure drop of pure R134a is larger than that of pure R12 at the same mass velocity. The pressure drop difference between R134a and R12 rises with an increase in the oil mass fraction, and that of R134a becomes about 1.4 times that of R12 at an oil mass fraction of 10%.

The pressure drop ratio, refrigerant-oil mixture versus pure refrigerant, of the R134a-PAG mixture is also larger than that of the R12-mineral oil mixture, as shown in Figure 7. It is considered that the difference of viscosity for the R134a-PAG mixture and the R12-oil mixture causes the pressure drop difference. The solubility of PAG to R134a is slight compared with mineral oil to R12 at the same temperature and pressure.

FORCED CONVECTION CONDENSATION HEAT TRANSFER COEFFICIENTS AND PRESSURE DROP

Experimental Apparatus

A schematic diagram of the apparatus is shown in Figure 8. This apparatus also consists of a refrigeration cycle and an oil circulating loop similar to the apparatus shown in Figure 1. However, the condenser is a double-pipe counterflow heat exchanger, and constitutes the test section by itself. The oil is eliminated from the compressor discharge gas by the oil separator. The refrigerant mass flow rates are controlled by using the expansion valve, and are measured with a turbine-type flow meter.

The oil contained in the compressor shell is taken out from the compressor shell bottom, and transferred to the test section inlet by a plunger pump. The oil mass flow rates are changed by controlling the plunger stroke, and are measured with a oil flow meter.

The test section consists of five subsections, each of which is a double-pipe counterflow heat exchanger, and is mounted horizontally in series. The outer tube of each is made of acrylic acid resin, and the inner tube is made of copper tubing 6.3 mm in outside diameter and 3.9 mm in inside diameter. Each section is 1 m long. The T-type (CC) thermocouples for measuring the tube wall temperature are attached at the tube wall at 90 degree distance and at 4 points along the length of the tube. The temperatures of the frigerant and water are also measured at the inlet and outlet of each subsection, and the refrigerant pressures are measured at the inlet and outlet of the test section.

Experimentation and Results

Testing was conducted by controlling the oil flow rate injected into the test section, while the refrigerant mass velocity and condensing pressure were maintained constant. The condensing pressure is controlled by a bypath valve mounted between the compressor outlet and inlet.

Inside wall temperatures are calculated from the outside temperatures. At the point in the subsections where the refrigerant-oil mixtures and water temperatures are not directly measured, they are calculated from the reading of the two temperatures at the inlet and outlet of the subsections assuming that the temperature changes linearly between these points. The average heat flux on the subsection is calculated from the heat energy absorbed by the water.

The influence of oil on the local heat transfer coefficients is shown in Figures 9 and 10 at a refrigerant mass velocity of 400kg/m²s. Figure 9 is the result for the R134a-PAG (polyalkylene glycol) mixture, and Figure 10 is for the R12-mineral oil mixture.

The oil mass fraction is defined by Equation 1, and the local heat transfer coefficient is given by Equation 2. The quality of the refrigerant-oil mixture is defined as the quality of the refrigerant contained in it, and calculated by Equation 3. In Equation 3, i_{rL} is the specific enthalpy of the refrigerant from the outlet of the test section at point L m, and is obtained from the energy equation:

$$M_r (i_{rL} - i_{ro}) + M_o c_{po} (t_{oL} - t_{oo}) = M_w c_{pw} (t_{wL} - t_{wo}) \quad (5)$$

The turbine-type flow meter measures the volume flow rates of the refrigerant-oil mixture, therefore, the mass flow rate of the refrigerant is obtained from the relations below:

$$V_{mix} = \left(\frac{G_r}{\rho_r} + \frac{G_o}{\rho_o} \right) \frac{\pi}{4} d_i^2 \quad (6)$$

Where the thermodynamic properties of R134a are calculated by using the data presented by Willson (1988).

As can be seen from Figure 9 and 10, with a decrease in quality the heat transfer coefficient of pure refrigerant decreases gradually because of the increase in condensing liquid film thickness. However, the heat transfer coefficients of refrigerant-oil mixtures increase in the high quality region and gradually decrease in the low quality region with a decrease in quality. In the high quality region, a thermal resistance to heat transfer depends on the thermal conductivity of the liquid refrigerant-oil mixture film which blankets the inner surface of the tube. With an decrease in quality, the refrigerant content increases in the film and the thermal conductivity of it increases, therefore, the heat transfer coefficients increase. However, as the quality decreases more, the film thickness increases and the heat transfer coefficients decline.

Figure 11 shows the influence of oil on the average heat transfer coefficient which is found by integrating the local heat transfer coefficients over the whole quality range. The heat transfer coefficient of pure R134a is larger than that of R12 by about 20%, because the thermal conductivity of liquid R134a is larger than that of R12. With an increase in the oil mass fraction, the average heat transfer coefficient of R134a decreases about 28% at an oil mass fraction of 6% and becomes the same value as R12.

Figure 12 shows the effect of the oil mass fraction on the pressure drops of R134a and R12 at a mass velocity of 400kg/m²s. The pressure drop of R12 increases gradually with an increase in the oil mass fraction and increases about 10% at an oil mass fraction of

10%. On the other hand, the pressure drop of R134a increases remarkably and becomes 1.6 times that of pure R134a at an oil mass fraction of 10%. The specific volume of R134a is larger than that of R12, therefore, the pressure drop of pure R134a is larger than that of pure R12 at the same mass velocity. The pressure drop difference between R134a and R12 rises with an increase in the oil mass fraction, and that of R134a becomes about 2.5 times that of R12 at an oil mass fraction of 10%. It is considered that the difference in viscosity for the R134a-PAG mixture and the R12-oil mixture causes the pressure drop difference. The solubility of PAG to R134a is slight compared with mineral oil to R12 at the same temperature and pressure.

CONCLUSIONS

The influence of oil on the heat transfer coefficients and pressure drop for forced convection and condensation of R134a and R12 are confirmed experimentally in the horizontal tube. The conclusions are summarized below.

(1) With an increase in the oil mass fraction, the boiling heat transfer coefficients of R134a decrease about 20% at an oil mass fraction of 10%.

(2) The deviation in heat transfer coefficients for R134a and R12 are small, and the maximum is 10% over the oil mass fraction range of 0 to 10%.

(3) The average pressure drop of R134a and R12 increases linearly with an increase in the oil mass fraction, and the difference between them increases and R134a becomes about 1.4 times that of R12 at an oil mass fraction of 10%.

(4) The condensation heat transfer coefficient of pure R134a is larger than that of R12 by about 20%. With an increase in the oil mass fraction the average heat transfer coefficient of R134a decreases about 28% at an oil mass fraction of 6% and, becomes the same as R12.

(5) The condensation pressure drop of R12 increases gradually with an increase in the oil mass fraction and increases about 10% at an oil mass fraction of 10%. On the other hand, the pressure drop of R134a increases remarkably and becomes 1.6 times that of pure R134a at an oil mass fraction of 10%.

(6) The pressure drop difference between R134a and R12 increases with an increase in the oil mass fraction, and R134a becomes about 2.5 times that of R12 at an oil mass fraction of 10%.

NOMENCLATURE

c_p = specific heat (J/kg·K)
d_i = inner diameter (m)
G = mass velocity ($kg/m^2 \cdot s$)
h = heat transfer coefficient ($W/m^2 \cdot K$)
h_{fg} = latent heat of vaporization (J/kg)
i_r = enthalpy (J/kg)
i_{is} = enthalpy of liquid refrigerant (J/kg)
L = length (m)
M = mass flow rate (kg/s)
Q_{sub} = heat flow generated by subsection heater (W)
q = heat flux (W/m^2)
t_o = oil temperature (K)
t_w = water temperature (K)
V_{mix} = volume velocity of mixture (m^3/s)
W = perimeter length (m)

x = quality
ΔP = pressure drop (kPa)
ξ_o = oil mass fraction
π = circular constant
ρ = density (kg/m^3)

Subscripts

A = average
L = point L
o = oil
r = refrigerant
W = water

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Wilson, D. P., and Basu, R. S. 1988. "Thermodynamic properties of a new stratospherically safe working fluid-refrigerant 134a." ASHRAE Transactions, Vol. 94, Part 2

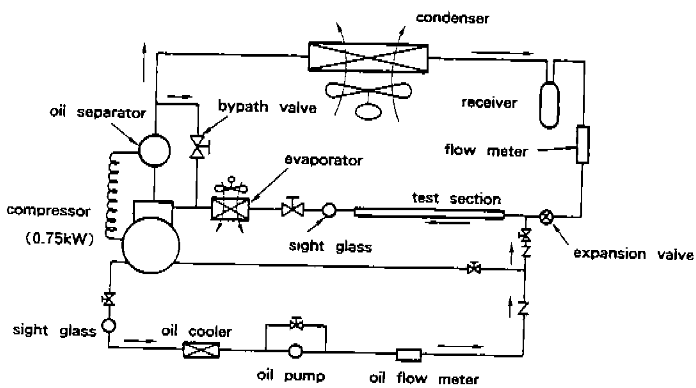

Figure 1 Schematic diagram of apparatus

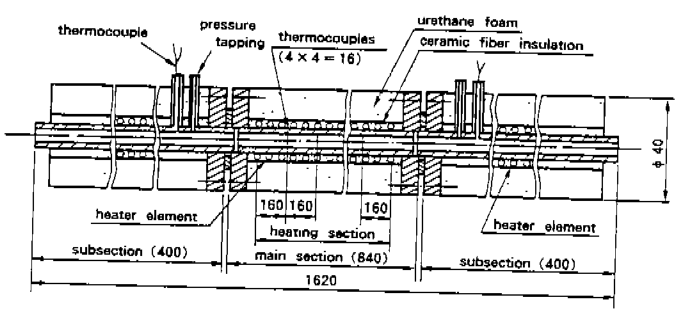

Figure 2 Test section

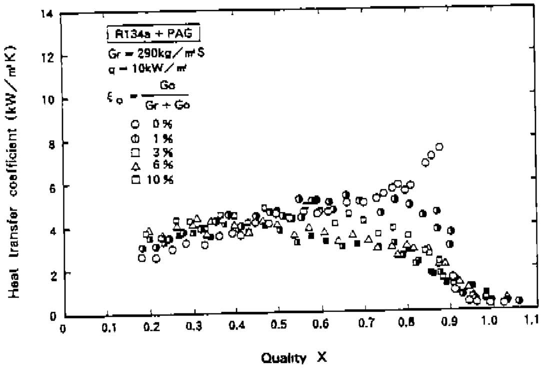

Figure 3 Influence of oil on boiling heat transfer coefficients (R134a)

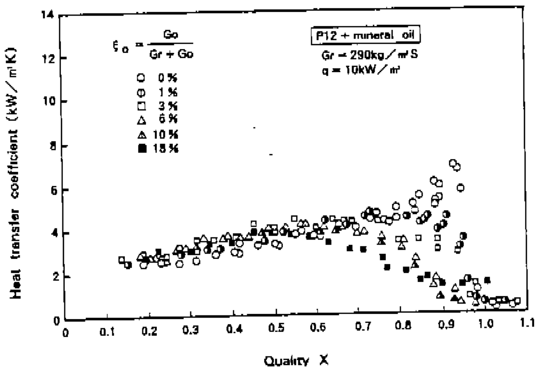

Figure 4 Influence of oil on boiling heat transfer coefficients (R12)

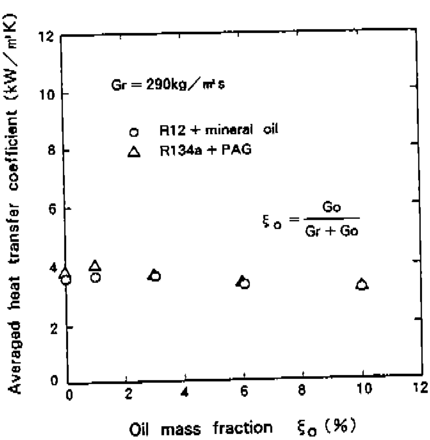

Figure 5 Influence of oil on boiling heat transfer coefficients

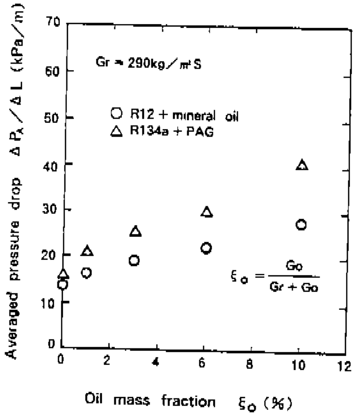

Figure 6 Effect of oil mass fraction on pressure drop

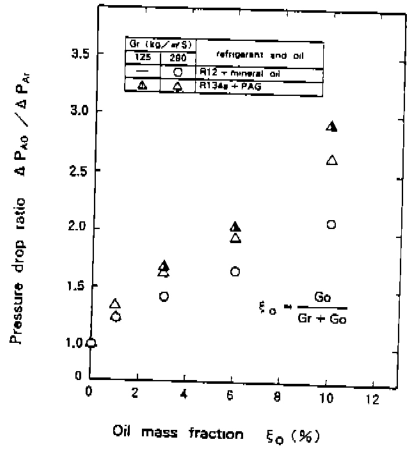

Figure 7 Effect of oil mass fraction on pressure drop ratio

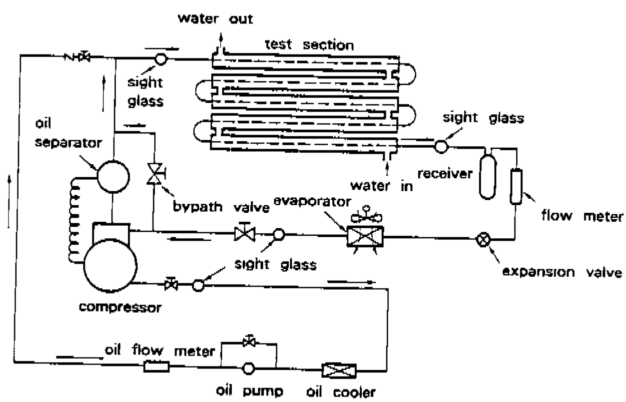

Figure 8 Schematic diagram of apparatus

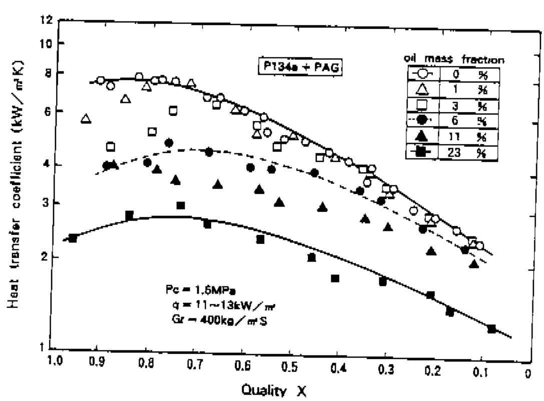

Figure 9 Condensation heat transfer coefficients (R134a)

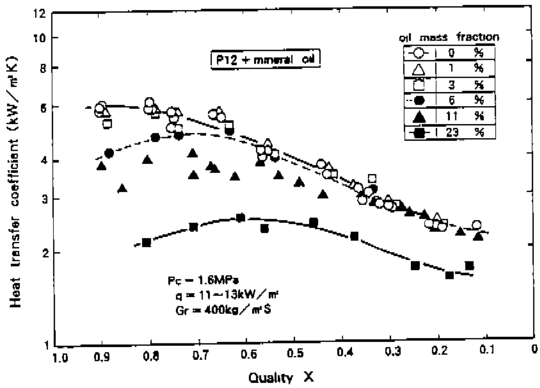

Figure 10 Condensation heat transfer coefficients (R12)

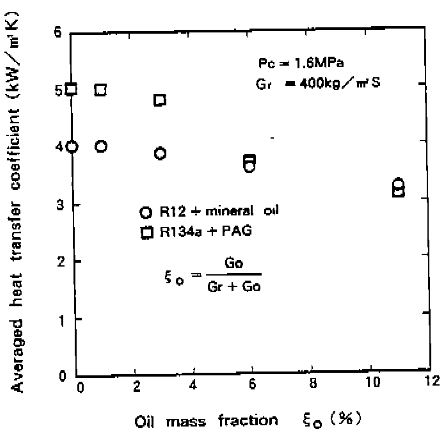

Figure 11 Influence of oil on condensation heat transfer coefficients

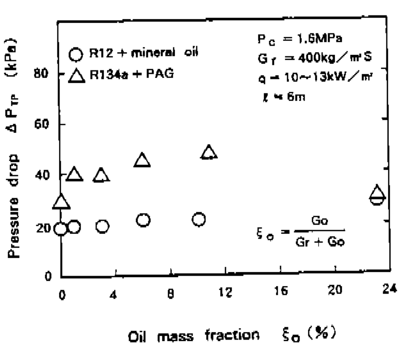

Figure 12 Influence of oil on pressure drop